LIQUID-FILM HEAT-TRANSFER COEFFICIENTS IN FORCED-CIRCULATION EVAPORATORS

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Temperature Drops and Liquid-Film Heat Transfer Coefficients in Vertical Tubes[1]

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THE design of forced-circulation evaporators is at present accomplished by the use of over-all coefficients which are mainly a matter of experience. It has been long realized that the proper approach to this problem is through the use of film coefficients, but there are practical difficulties.

The simplest equation for the design of an evaporator is:

$$Q/\theta = UA\,\Delta T_T \qquad (1)$$

ΔT_T is the true mean temperature drop based on the true mean liquid temperature and the steam temperature and U is the over-all coefficient of heat transfer defined as

$$U = \frac{1}{\frac{1}{h_S} + \frac{k}{L_W} + \frac{1}{h_L}} \qquad (2)$$

This equation holds rigidly only for the case where no thermal resistances are present except the steam film, the wall of the tube, and the liquid film. Where other resistances are present, such as scale and impure steam, they must be included by additional terms in Equation 2. This definition also assumes that the areas of the steam film and the liquid film are appreciably equal.

Equation 1 cannot be used because ΔT_T, the true mean temperature drop, is not known. Most designs are based on the over-all apparent temperature drop because it can be readily determined. The over-all apparent temperature drop, ΔT_A, is defined as the temperature difference between the steam in the steam jacket and vapor in the evaporator body. Coefficients based on this definition are known as over-all apparent coefficients of heat transfer. These coefficients are entirely fictitious because the effect of ignoring boiling point elevation, circulatory system heat losses, temperature changes of the liquid in the tubes, etc., is to show a lower coefficient than the true over-all coefficient of heat transfer. Nevertheless, a large body of data has been built up on this system which has been useful in design. On this basis, design has been a matter of experience. It is desirable to reduce this operation to a logical basis by the use of film coefficients. From the film coefficients can be calculated the over-all coefficient of heat transfer, U, by the use of Equation 2.

[1] Complete tables of data obtained in these experiments will be found in the *Transactions of the American Institute of Chemical Engineers.*

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The temperature drop must be the true mean temperature drop.

Equation 1 cannot be used until the apparent over-all temperature drop from steam to vapor and the average true temperature drop from steam to average liquid in the tube can be reconciled, and until means of calculating the film coefficients without the necessity of knowing the tube wall temperature are available.

Many investigators have reported that the average true liquid film coefficient of heat transfer for the simple heating of fluids moving in forced convection inside of tubes could be predicted by use of the Dittus and Boelter relation (*4*):

$$\frac{hD}{k} = a\left(\frac{Du\rho}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{0.4} \qquad (3)$$

Here h, the average true liquid film coefficient of heat transfer, is shown to vary with some function of the tube diameter, the velocity, the physical properties of the fluid evaluated at average bulk temperature, and the constant a.

McAdams (*10*) discussed this equation fully. He concluded that Equation 3 is the best general equation for the simple heating of fluids, with the constant a equal to 0.0225, where all units are consistent.

Logan, Fragen, and Badger (*9*), working on a semicommercial forced circulation evaporator with 8-foot tubes under essentially nonboiling conditions, found that the Dittus and Boelter equation correlated their data. It would appear, then, that the applicability of this equation has been proved in the case where appreciable boiling does not occur.

It is important, therefore, to the problem of the logical design of forced-circulation evaporators that the relation between the over-all apparent temperature drop and the true mean temperature drop be known. Further, the effect of appreciable boiling upon the liquid film coefficient of heat transfer in such a system should be examined. This paper presents the results of such investigations made on a single vertical-tube forced-circulation evaporator. The steam film characteristics of the same runs made on the same apparatus were reported in a previous paper (*11*).

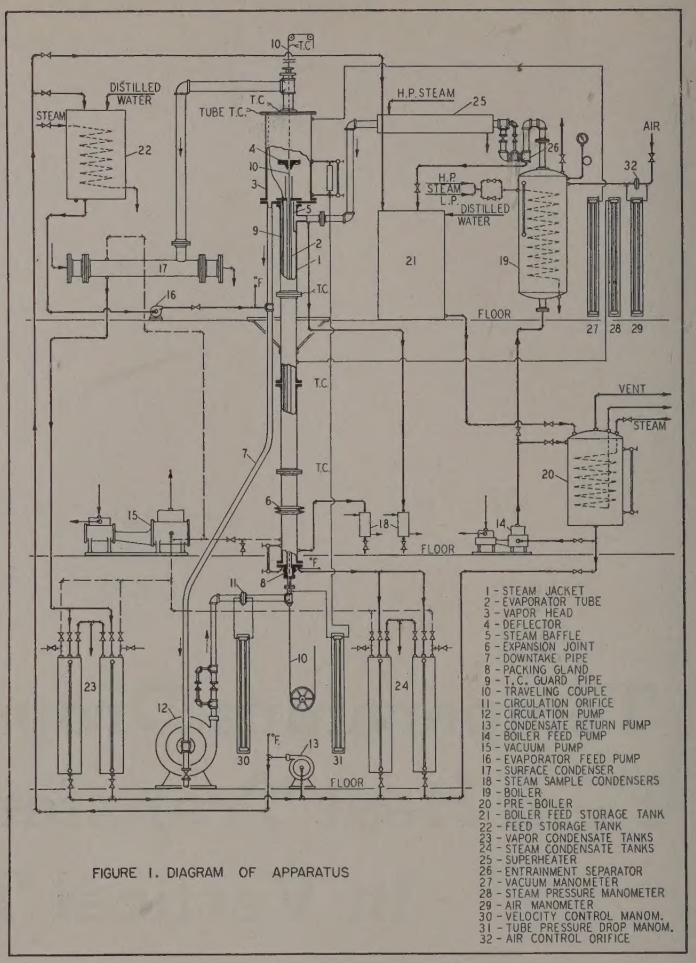

FIGURE 1. DIAGRAM OF APPARATUS

Literature Survey

McAdams (*10*) reviewed the work done on the simple heating of water and other fluids both in horizontal and vertical tubes. The data on the heat transfer to boiling fluids in forced convection are very few. Logan, Fragen, and Badger (*9*) had some boiling when using a sugar solution in a forced-circulation evaporator with twelve 8-foot copper tubes, 0.75 inch inside diameter. Their results followed the Dittus and Boelter equation.

Claasen (*1*), in 1902 in a theoretical discussion, stated that vapor bubbles clinging to the heating surface should reduce the coefficient of heat transfer by blanketing off part of the active area. Jakob and Fritz (*7*) verified this hypothesis, using a horizontal plate evaporator. Cleve (*2*), Linden and Montillon (*8*), and Cryder and Gilliland (*3*) took data on water boiling under natural convection in vertical, inclined, and horizontal tubes, respectively. These investigations, with the exception of that of Logan, Fragen, and Badger, are not significant to the present work.

No work on the correlation of the over-all apparent temperature drop with the true mean temperature drop has been reported.

Experimental Apparatus and Procedure

The experimental apparatus was described in a previous paper (*11*):

It was essentially a single-tube forced-circulation evaporator. The tube was 12 feet long, 0.76 inch inside diameter, with an inside heating surface of 2.367 square feet. The arrangement of the equipment is shown in Figure 1.

All data were obtained on the transfer of heat to distilled water. The variables and the ranges covered by each were: apparent boiling point of the liquid, 140° to 212° F.; condensing steam temperature, 176° to 248° F.; inlet velocity of liquid, 2.5 to 15 feet per second; apparent over-all temperature drop, 18° to 72° F.; and percentage of evaporation, 0.25 to 5.00, approximately.

Condensate from the steam space was measured in closed calibrated receivers. From the weight of this condensate and the known temperature and pressure of the steam, the total heat input was calculated. A semiquantitative check was obtained by measuring the condensate obtained from the vaporization of the liquid in the evaporator tube. Entrainment prevented a close check. The evaporator was completely lagged against heat loss.

Temperatures along the tube were measured by ten copper-constantan thermocouples installed at 15-inch intervals by the method of Hebbard and Badger (*5*). The liquid temperature rise in the tube was followed by means of a traveling copper-

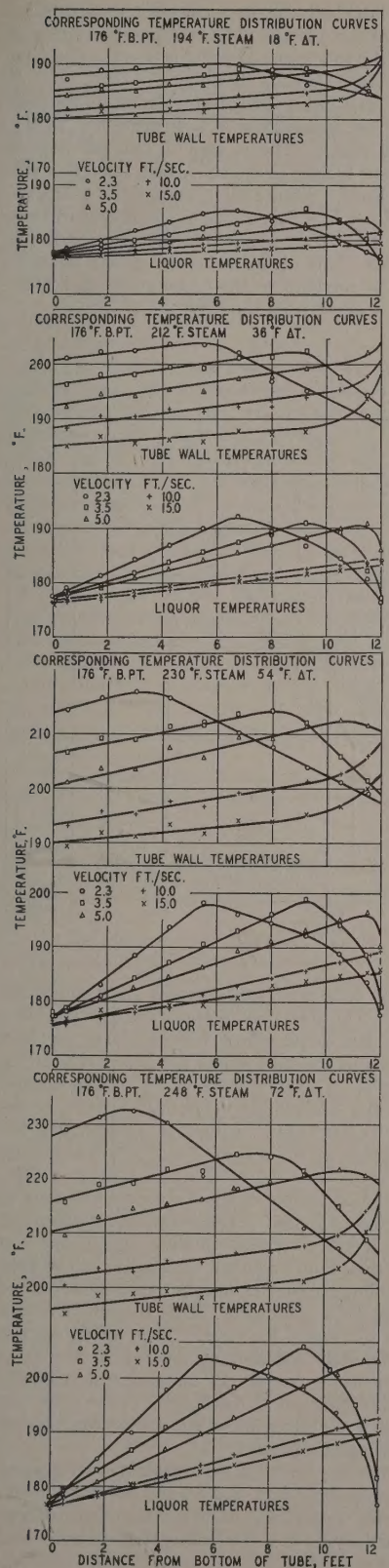
CORRESPONDING TEMPERATURE DISTRIBUTION CURVES
176 °F. B.P. 194 °F. STEAM 18 °F. ΔT.
TUBE WALL TEMPERATURES
LIQUOR TEMPERATURES
VELOCITY FT./SEC.
2.3 + 10.0
3.5 × 15.0
5.0
TEMPERATURE, °F.
CORRESPONDING TEMPERATURE DISTRIBUTION CURVES
176 °F. B.P. 212 °F. STEAM 36 °F. ΔT.
CORRESPONDING TEMPERATURE DISTRIBUTION CURVES
176 °F. B.P. 230 °F. STEAM 54 °F. ΔT.
CORRESPONDING TEMPERATURE DISTRIBUTION CURVES
176 °F. B.P. 248 °F. STEAM 72 °F. ΔT.
DISTANCE FROM BOTTOM OF TUBE, FEET

Figure 2. Temperature Distribution of Tube Walls and Liquids

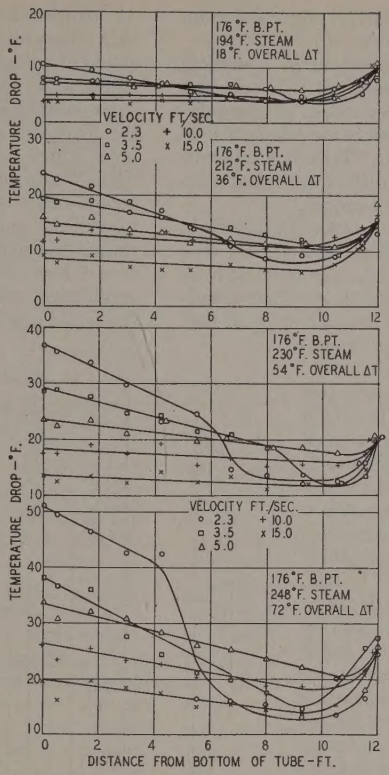

Figure 3. Temperature Drop across Liquid Film

constantan thermocouple. The lead wires were carried out of the evaporator through a $^1/_8$-inch outside diameter brass tube which moved through a packing gland. The junction was soldered direct to the end of the tube, and three brass fins were attached an inch above the hot junction to keep the thermocouple centered in the evaporator tube. A copper wire of approximately the same diameter was soldered to the end of the small tube to act as a tag wire in keeping the brass tube taut in the evaporator tube. By a pulley and take-up system the junction of the thermocouple could be moved to any predetermined point in the evaporator tube. With this arrangement small resistance was offered to the flow of liquid in the tube, and the resistance was constant no matter where the junction was.

Other thermocouples were used to measure the temperature of the steam jacket, vapor space above the tube, liquor inlet at the bottom of the tube, and superheated steam. All readings were made on a Leeds & Northrup type K-2 potentiometer.

The temperature of the liquid side of the tube was calculated from the tube couple readings, the depth to which their junctions were imbedded in the tube wall, and the temperature gradient corresponding to the known rate of heat flow through the tube wall. Typical curves showing the corresponding temperature distributions of tube wall temperatures and liquid temperatures are represented by Figure 2. From these data Figure 3 was drawn to show the temperature drop over the liquid film at any point on the tube. All the curves were similar in their characteristics; those shown were from four representative runs.

The length of the runs was 40 minutes. During that time manometer readings were taken every 2 minutes. All thermocouples were read at least three times during a run. The levels of the condensate in the vapor and steam condensate tanks were read at frequent intervals and plotted to show constancy of operation.

Liquid-Film Coefficients for Long Vertical Tubes

Since Logan, Fragen, and Badger found that they could correlate their data reasonably well by means of a Dittus and Boelter type equation, that was the first step taken in the present work (Equation 3).

The Prandtl group, $c\mu/k$, was not a primary variable in this work. Therefore, the experience of past workers in the general field was drawn upon, and this group was introduced raised to the 0.4 power.

The Reynolds and Nusselt numbers, $Du\rho/\mu$ and hD/k, were calculated from the observed data. In these calculations consistent units were used throughout. The only deviation from past work occurred in the use of the data of Schmidt and Sellschopp (*16*) for the values of the thermal conductivity, k. Their results are lower for water approaching 212° F. than the thermal conductivities based on the work of Jakob (*6*). Since this investigation was on water at temperatures from 140° to higher than 212° F., and since quantity k entered into the Dittus and Boelter equation to the 0.6 power, the calculated values of this work were lower than if Jakob's data had been used. This difference varied from 0 per cent water at 151° F. to 3.5 per cent for water at 212° F.

To check the power function of the Reynolds number, the quantity $\left(\frac{hD}{k}\right)\Big/\left(\frac{c\mu}{k}\right)^{0.4}$ was plotted against $(Du\rho/\mu)$. This is shown as Figure 4. A line with a slope of 0.8 on the logarithmic plot adequately correlates the data except at very low velocities where $Du\rho/\mu$ is less than 65,000. The median of the calculated values of constant a for the runs with a Reyn-

olds number higher than 65,000 was 0.0278, and the line on Figure 4 is drawn with a equal to this value.

For water flowing under forced convection in a vertical heated tube with a small degree of boiling, the Dittus and Boelter equation furnishes a satisfactory means of determining the true mean liquid film, h_L. This is a confirmation of the work of Logan, Fragen, and Badger. Equation constant a is, respectively, 0.0205 and 0.0278 for the former and present investigators. This difference is not serious.

For runs with appreciable boiling in the tube (in this work, those with velocities of 2.3 and 3.5 feet per second), the true mean liquid-film coefficient of heat transfer predicted by the Dittus and Boelter equation is much lower than experiment shows. With the data at hand, it was not found possible to give more than semiquantitative demonstrations of this fact.

An attempt was made to break up into two parts the calculations on the runs in which appreciable boiling occurred. Examination of the temperature distribution curves (Figure 2) shows that active boiling in the tube reduces the temperature of the liquid by the conversion of sensible heat to latent heat. If the point where this sudden drop in liquid temperatures occurs is taken as the point of active boiling, then heat transmission in the tube length can be considered in two sections. Simple heating occurs in the part of the tube up to the break in the curve. Thus in Figure 4 it can be assumed that 5.5 feet of the tube are operating under simple heating with 2.3 feet per second velocity; 9.2 feet with 3.5 feet per second; 11.5 feet with 5.0 feet per second; and practically the entire tube with higher velocities. If the heat quantity transmitted over these lengths is calculated on the basis of the pounds of circulation per hour times the sensible heat increase, then the mean true liquid film coefficient can be determined from the Newton equation:

$$Q/\theta = h_L A_L \Delta t_L \qquad (4)$$

All temperature plots had the same characteristics. The only differences were those of amount of temperature rise and extent of boiling in the tube caused by changes in the effective temperature of the steam outside the tube.

Examination of any liquor temperature plot shows that the temperature of the liquid rises in a straight line to the point where boiling starts. The 10 and 15 foot per second velocities do not show boiling in the tube, and the 5 foot per second shows very little. After boiling starts, the temperature of the water drops in a curved line to a temperature substantially equal to that of the vapor in the evaporator, in the case of low velocities. The higher velocities do not come to equilibrium with the pressure in the vapor space until after the liquid has been ejected from the tube.

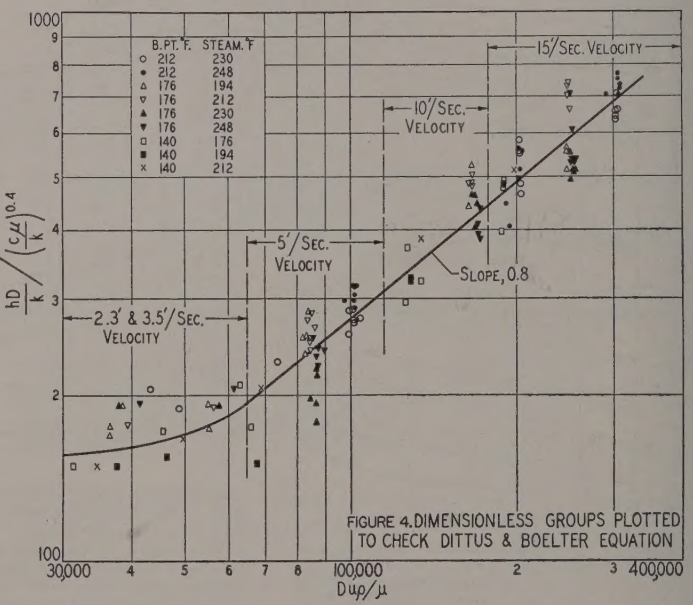

FIGURE 4. DIMENSIONLESS GROUPS PLOTTED TO CHECK DITTUS & BOELTER EQUATION

The temperature drop over the liquid film, Δt_L, is the integrated temperature difference from liquid to the tube wall thermocouples corrected for the resistance of the portion of the tube wall from the thermocouple junction to the inside surface of the tube.

Inserting this calculated coefficient in the Dittus and Boelter equation, the constant a for the nonboiling section was calculated. The average constant, 0.0293, is much closer to the constant, 0.0278, for runs entirely nonboiling, than would be expected from this procedure.

The value of the Nusselt criterion for the boiling portion is calculated by Newton's equation from the heat transferred over the entire tube, less that assigned to the nonboiling section. The average value of the coefficient to boiling water is approximately twice that of the coefficient to nonboiling water. The temperature drop relations determined by this method exhibit some inconsistencies.

Temperature Distributions Observed

Reference has been made to plots of the temperature distribution and temperature difference between the tube wall and the liquid. These plots have interesting ramifications.

A description of the method of attaching the thermocouples to the tube wall has been described by Hebbard and Badger (*5*). The traveling, or liquid, thermocouple is described above. Figure 2 is presented as representative of all the runs.

The plots of the tube wall temperatures show much the same phenomena. The tube wall temperature also rises in a straight line, but the break where boiling starts precedes somewhat the point in the tube where the liquid showed its change in the curve. The tube wall, where boiling occurs, loses temperature in a straight line in contradistinction to the curved line taken by the water. Somewhat surprisingly, the tube wall temperature at the top of the tube in nonboiling runs up sharply.

In the plots of the Δt_L values of these same runs (Figure 3) an interesting similarity is seen. The temperature difference from tube wall to liquid, for the nonboiling runs, drops in a straight line to a minimum and then rises again. The runs in which there is boiling are curved lines, or a mixture of curved and straight lines, to a minimum. This minimum occurs about 2 feet from the top of the tube. Then all curves rise and come approximately together at the top of the tube. That is, Δt_L for the same over-all ΔT but different velocities may be considerably different at the bottom of the tube, but they all go through a minimum about 2 feet from the top and they all rise to a constant value.

The influence of boiling on Δt_L is very marked in the graphs. Based on the temperature data and on the observed coefficients, the following hypothesis of the mechanism of heat transfer to liquids flowing in forced convection in a vertical tube evaporator is suggested.

Mechanism of Heat Transfer to Liquids in a Forced-Circulation Evaporator

The liquid enters the tubes of a forced-circulation evaporator at substantially the equilibrium temperature in the vapor space, although in some cases it may be somewhat cooler than this temperature owing to heat losses in the return line and the use of cold make-up feed. Heat is transferred through the tube wall and superheats the liquid above the equilibrium temperature in the vapor space. The liquid is kept from vaporizing by the hydrostatic and friction heads above it. In this region the Dittus and Boelter equation holds.

The Dittus and Boelter equation is based on the Nusselt theory (*14*) of heat transfer to liquids in pipes, based on conduction and convection. The heat must pass by conduction through a viscous film and by convection through the bulk of the liquid. On the outside of the tube, heat passes from condensing steam to the tube by two mechanisms—film condensation (either viscous or turbulent) and dropwise condensation. Schmidt, Schurig, and Sellschopp (*15*) and Nagle and co-workers (*12*, *13*) showed that dropwise condensation can produce coefficients four to eight times that of viscous film condensation. If producing a bubble of water from steam can be so much more effective than transferring heat through a viscous film of condensate on the outside of the tube, then it does not lie beyond possibility that the production of a bubble of steam in the inside of the tube would likewise be a better heat transmission agency than a viscous film. The minute drop of steam so formed is removed immediately, and convection into the cooler bulk of water condenses it. Such mechanism would account for the uniformly higher coefficients obtained than those the Dittus and Boelter equation predicts.

As the superheated water passes up the tube, it becomes still hotter. This has the effect of raising both the Prandtl and the Reynolds numbers. Consequently, the heat transferred through the viscous film increases because of scouring effects. This is shown by the drop in Δt_L shown by the temperature difference curves (Figure 3) as the liquid proceeds up the tube.

At the point where marked boiling occurs in the tube, the hydrostatic and friction heads are now not sufficient to prevent vaporization of the water. An evolution of stable bubbles occurs which tremendously increases the volume of the mixed liquid and vapor. This induces a large velocity which reduces the thermal resistance of the liquid film over the part of the tube in boiling action. The liquid film coefficient for the boiling sections was approximately twice that of the nonboiling section for these runs.

The lag in the maximum temperature of the liquid as compared to the tube wall loses most of its significance when checked against the corresponding Δt_L. The temperature difference drops smoothly with increasing heat input. This indicates that vaporization in the tube is not sudden but progressive action. The temperature of the tube wall and liquid are not independent variables but are dependent on Δt_L much as the bodies of a multiple-effect evaporator adjust their own temperatures; hence the tube wall and liquor temperature must be in such relationship as to show a smoothly decreasing Δt_L.

Near the top of the tube all temperature drops over the liquid film approach a constant value, no matter what inlet velocity is used. This requires the tube wall temperature of the nonboiling runs to rise. The meaning of this is obscure.

To summarize this hypothesis of the mechanism of heat transfer in a vertical-tube forced-circulation evaporator proposes a three-component action: (1) heat transfer through a viscous liquid film; (2) the formation of incipient bubbles of steam that are reabsorbed in the cooler liquid just the same as those that can be observed by watching a pan of water over a burner; (3) the formation of stable bubbles by flash evaporation of the superheated liquid. If this hypothesis is true, the determination of coefficients to boiling liquids becomes a matter of degree of the relative predominance of the three actions. Possibly the best correlation of this mixed mechanism would be on the basis of the percentage of liquid evaporated.

True and Apparent Temperature Drops in a Forced-Circulation Evaporator

In the logical design of a forced-circulation evaporator, it has been shown that the equation,

$$Q/\theta = UA\,\Delta T_T \qquad (1)$$

cannot be used at present because of the uncertainties entering into the over-all coefficient of heat transfer, U, and because there was no simple way of determining the true mean temperature drop, ΔT_T.

Where appreciable boiling does not occur in the tubes (i. e., the usual case), the liquid film coefficient, h_L, can be predicted by the Dittus and Boelter equation. The steam film coefficient does not usually control, especially where dropwise condensation occurs. Other thermal resistances which go to make up the over-all coefficient, U, can be predicted or avoided. The bar to use of Equation 1 is not, therefore, one of coefficients but of temperature drops.

If from the over-all apparent temperature drop the true mean temperature drop can be calculated, then a logical design of a forced-circulation evaporator may be made. This means that some relation must be found between the temperature indicated by the pressure of the vapor in the evaporator body and the average temperature of the liquid. The steam temperature is common to both methods of defining temperature drops.

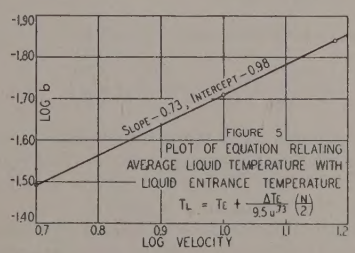

FIGURE 5
PLOT OF EQUATION RELATING AVERAGE LIQUID TEMPERATURE WITH LIQUID ENTRANCE TEMPERATURE

$$T_L = T_C + \frac{\Delta T_E}{9.5^{0.73}} \left(\frac{N}{12}\right)$$

In the simplest case (where there is no elevation of boiling point and no temperature loss in the circulatory system, and where the make-up feed is preheated) the temperature of the vapor is the same as the temperature of the liquid entering the tubes or

$$T_V = T_E \qquad (5)$$

The question of the average liquid temperature when there is no marked boiling in the tubes can be answered by considering the temperature distribution curves.

Figure 2 shows that the temperature of the liquid in the 5, 10, and 15 foot per second velocity runs rises in a straight line as the top of the tube is approached. These lines have increasing slope with increasing ΔT and decreasing slope with increasing velocity.

For each velocity the slopes of the lines were calculated and averaged. By plotting the log of the slopes against the log of the velocities, an equation of the following type was found to fit the data:

$$T_X = T_E + b(u)^s \Delta T_E(X) \quad (6)$$

where T_X = temp. of liquid at any point, ° F.
T_E = temp. of entering liquid, ° F.
u = velocity, ft./sec.
ΔT_E = temp. drop from steam to entrance temp. of liquid, ° F.
X = distance from bottom of tube, ft.
s, b = constants

On Figure 5 the constants are evaluated. The slope, s, is -0.73 and the intercept, b, is 1/9.5. The average liquid temperature for nonboiling runs lies halfway up the tube; therefore

$$T_L = T_E + \frac{\Delta T_E}{9.5(u)^{0.73}}\left(\frac{N}{2}\right) \quad (7)$$

where T_L = av. liquid temp. in tube

This is purely an empirical formula that describes the present data. It also fitted the data of Logan, Fragen, and Badger on runs taken at random, even though their liquid was strong sugar solution.

When concentrating a delicate solution that may be injured by high temperatures, the maximum temperature in the system may possibly be checked with the same equation, using the full length of the tube instead of the half-length.

The average true temperature drop is:

$$\Delta T_T = T_S - T_L \quad (8)$$

where ΔT_T = av. true ΔT
T_S = steam temp.
T_L = av. temp. of liquid in tube

Hence $$\Delta T_T = T_S - T_E - \frac{\Delta T_E}{9.5(u)^{0.73}}\left(\frac{N}{2}\right) \quad (9)$$

ΔT_E is equal to the apparent over-all temperature drop, ΔT_A, in this case.

Now $$\Delta T_A = T_S - T_V \quad (10)$$

where ΔT_A = apparent over-all temp. drop
T_V = vapor temp.

Therefore, from Equations 5, 9, and 10:

$$\Delta T_T = \Delta T_A - \frac{\Delta T_A}{9.5(u)^{0.73}}\left(\frac{N}{2}\right) \quad (11)$$

Thus the average true temperature drop is expressed in terms of the over-all temperature drop. This equation gives results about 1° to 2° F. high for ΔT_T in this investigation. No reason for the discrepancy is known.

With a solution having an appreciable elevation in boiling point, the above equation must be changed somewhat. In this case the entrance temperature of liquid to the tubes is the vapor temperature plus the boiling point elevation. The ΔT_E value of Equation 9 is $\Delta T_A - \Delta T_{BPE}$; then

$$\Delta T_T = \Delta T_A - \Delta T_{BPE} - \frac{\Delta T_A - \Delta T_{BPE}}{9.5(u)^{0.73}}\left(\frac{N}{2}\right) \quad (12)$$

Wherever there is a temperature loss in the circulatory system between the evaporator body and entrance to the tubes, or cold feed is introduced, the proper correction should be made on the temperature of the liquid entering the tubes.

These equations are nothing more than a temperature-drop balance and can be expressed in the general case as

$$\Delta T_A = \Delta T_T + \Delta T_H - \Delta T_F - \Delta T_R + \Delta T_{BPE} \quad (13)$$

where ΔT_A = apparent over-all ΔT
ΔT_T = av. true ΔT
ΔT_H = av. temp. rise of liquids in tubes, $\frac{\Delta T_E}{9.5(u)^{0.73}}\left(\frac{N}{2}\right)$
ΔT_F = cooling effect of cold make-up feed
ΔT_R = temp. loss through circulatory system
ΔT_{BPE} = boiling point elevation

Therefore, using only those terms which appear in the specific case and which are either known or easily calculated, Equation 1 becomes

$$Q/\theta = UA(\Delta T_A - \Delta T_H + \Delta T_F + \Delta T_R - \Delta T_{BPE} \ldots) \quad (14)$$

and the rational design of forced-circulation evaporators awaits only a confirmation of the above equations for liquids other than water and sugar solutions.

Summary

1. In investigating the average true liquid-film heat transfer coefficients of a long vertical-tube forced-circulation evaporator operating with velocities and temperature drops that do not cause marked boiling in the tubes, Logan's claim that an equation of the form of the Dittus and Boelter relation adequately correlates the data is confirmed. This demonstrates that the predominant mechanism in the usual forced-circulation evaporator is simple heating and is in complete accord with the work of Logan, Fragen, and Badger who employed a shorter tube.

The equation which best correlated the results on nonboiling water in this investigation is

$$\frac{hD}{k} = 0.0278\left(\frac{Du\rho}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{0.4} \quad (15)$$

2. Semiquantitative determinations of the liquid-film coefficient to boiling water lead to qualitative conclusions that this coefficient is much greater than would be predicted by the above equation based on predominately nonboiling conditions.

3. Film temperature drop and distribution curves are regular and uniform for all cases. They form a firm basis upon which interpretation of the essential heat transfer mechanisms can be approached with confidence.

Based on the temperature and heat transfer data, a hypothesis of heat transfer in a vertical-tube forced-circulation evaporator is presented. This hypothesis visualizes a three-part mechanism of simple heating through viscous films, incipient vaporization, and stable vaporization by conversion of sensible heat to latent heat.

4. Based on the temperature distribution curves, a formula relating the average liquid temperature in evaporator tubes to the entrance temperature is advocated:

$$T_L = T_E + \frac{\Delta T_E}{9.5(u)^{0.73}}\left(\frac{N}{2}\right)$$

This equation should be used in evaluating the physical properties of the liquid for the Dittus and Boelter type equation, especially where a high temperature drop is being used. As is customary in forced-circulation evaporator design, the velocity of the liquid should be sufficient to prevent marked boiling in the tubes.

5. With the means at hand to determine the average temperature increase of the liquid in the evaporator tubes,

$$\Delta T_H = \frac{\Delta T_E}{9.5(u)^{0.73}}\left(\frac{N}{2}\right)$$

The reconciliation of the apparent over-all temperature drop with the average true temperature drop follows very simply. The average true temperature drop is equal to the apparent over-all temperature drop less the average temperature rise of the liquid in the tubes less the boiling point elevation plus the temperature loss in the circulatory system plus the cooling effect of cold make-up feed.

This is an essential step in the development of a method of rational design of forced-circulation evaporators.

Nomenclature

A = area of heating surface, sq. ft.
a = constant in Dittus and Boelter equation
b = constant in liquid temp. rise equation
h = true mean film coefficient of heat transfer
k = thermal conductivity
L_W = tube wall thickness, ft.
N = length of tube, ft.
s = constant in liquid temp. rise equation
u = liquid velocity, ft./sec.
Q/θ = heat transferred, B. t. u./hr.
T = temp., ° F.
Δt = temp. drop across film
$Du\rho/\mu$ = Reynolds criterion, consistent units
$c\mu/k$ = Prandtl criterion, consistent units
hD/k = Nusselt criterion, consistent units

Subscripts:

A = apparent over-all temp. drop
T = true over-all temp. drop
S = steam
V = vapor in evaporator
L = liquid at mean temp.
E = liquid at entrance temp.
H = temp. rise of liquid in tube
BPE = boiling point elevation
F = evaporator make-up feed
R = temperature loss from system

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